Automobile Gearbox Design Assignment

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Selected Car: [2024 FORD MUSTANG - Getrag MT-82]
Link for all the images attached - [This folder]

1. Introduction & Objectives

Designing a manual gearbox is a challenge in itself. It requires consideration of torque capacity, gear ratio selection, bearing life, lubrication, thermal management, manufacturability, and long-term reliability. This report focuses on the design of a 6-speed manual gearbox derived from the 2024 Ford Mustang's Getrag MT-82 transmission.

2. Core Vehicle & Transmission Data

1. **Key Vehicle & Powertrain Data**
    a. Engine Peak Torque: 563 Nm @ 4900 RPM
    b. Max Power: 480 HP (≈ 358 kW) @ 7150 RPM
    c. Max Engine RPM (Redline): 7500 RPM
2. **Transmission Ratios (MT-82, 6-speed manual):**
    a. 1st: 3.237
    b. 2nd: 2.104
    c. 3rd: 1.422
    d. 4th: 1.000
    e. 5th: 0.814
    f. 6th: 0.622
3. **Final Drive Ratio:** 3.73:1 (TORSEN LSD)
4. **Tire Size (Rear):** 275/40R19; diameter ≈ 703 mm ⇒ circumference ≈ 2.21 m
5. **Approx. Vehicle Mass:** 1740 kg
6. **Typical Real Top Speed:** 250 km/h (due to aerodynamic drag and power limits)

3. Theoretical Calculations for Speed and Torque (Ideal vs Realistic)

1. **Using the formula on the gear ratios given**

$$n_{\text{wheel},\,i} = \frac{n_{\text{engine}}}{(\text{Gear Ratio}_i) \times (\text{Final Drive})},$$

$$V_{\text{vehicle},\,i} = n_{\text{wheel},\,i} \times (\text{Tire Circumference}) \times \frac{60}{1000}.$$

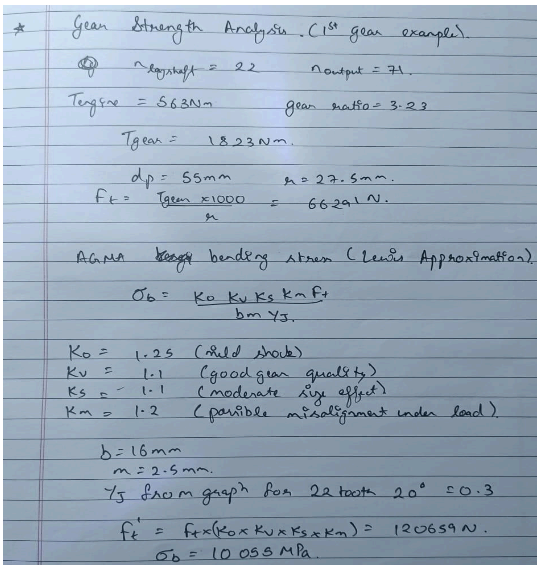
Where n_{engine}= 7500 RPM(Redline) and the Tire Circumference = 2.21m

2. **Ideal vs realistic Limit of each gear**:

Gear	Gear Ratio	Ideal Limit	Realistic Limit	Engine RPM at Realistic Limit
1st	3.237	82.2 km/h	82.2 km/h	620 rpm
2nd	2.104	127.4 km/h	127.4 km/h	960 rpm
3rd	1.422	187.0 km/h	187.0 km/h	1410 rpm
4th	1.000	266.5 km/h	~250 km/h	2011 rpm
5th	0.814	327.2 km/h	~250 km/h	2467 rpm
6th	0.622	429.4 km/h	~250 km/h	3237 rpm

Due to the aerodynamic constraints the vehicle cannot go beyond 250 km/h therefore we observe that the vehicle has the potential to reach **top speed at 4th gear**(which we will discuss in the next segment).

3. **Gear Strength Analysis:**



* Gear Strength Analysis. (1^{st} gear example).

$n_{layshaft} = 22$ $n_{output} = 71$.

$T_{engine} = 563 Nm$ gear ratio = 3.23

$T_{gear} = 1823 Nm$.

$d_p = 55mm$ $r = 27.5mm$.

$$F_t = \frac{T_{gear} \times 1000}{r} = 66291 N.$$

AGMA ~~[illegible]~~ bending stress (Lewis Approximation).

$$\sigma_b = \frac{K_o K_v K_s K_m F_t}{b m Y_J}.$$

$K_o = 1.25$ (mild shock)
$K_v = 1.1$ (good gear quality)
$K_s = 1.1$ (moderate size effect)
$K_m = 1.2$ (possible misalignment under load).

$b = 16mm$
$m = 2.5mm$.
Y_J from graph for 22 tooth $20° = 0.3$

$F_t' = F_t \times (K_o \times K_v \times K_s \times K_m) = 120659 N$.
$\sigma_b = 10055 MPa$.

Observations:

1. A raw bending stress of around **10,000 MPa** appears extremely high by conventional standards, exceeding the strength of most steels - But this is mostly due to the assumption that only one tooth is taking all the load. When in reality there can be many scenarios like:
    a. **Load Sharing between multiple gears:**: In practice, there may be 2–3 teeth sharing the load at any moment. If 2 teeth share load equally, the per-tooth load drops by half, etc.

b. **Peak Engine Torque vs. Real Operating Condition**: Peak torque is not always present at redline. The stress peak might be lower or occur over short durations.

2. Accounting for multiple teeth in contact and advanced gear steels, the effective stress could become more realistic (e.g., 1500–3000 MPa range), still high but possibly within the capabilities of well-designed, heat-treated gears.

4. Why Do We Have 5th and 6th Gear if Max Speed Can Be Reached in 4th?

Often, high-performance cars might mechanically achieve maximum speed in 4th or 5th gear, but still include even taller gears. The reasons include:

1. **Fuel Efficiency and Lower NVH (Noise, Vibration, Harshness):** At highway speeds (e.g., 110–130 km/h), a 5th or 6th gear brings engine RPM down, reducing fuel consumption and noise. Modern emissions/fuel economy standards encourage cruising at lower RPM to cut CO2 emissions.
2. **Engine Wear and Comfort:** Keeping the engine at a lower RPM under moderate loads is beneficial for engine longevity and driver comfort. The lower the revs, the less wear and the quieter the cabin.
3. **Powerband Matching:** Depending on the engine's torque curve, 4th gear might produce enough power to overcome drag up to a certain speed, but 5th could be used for slightly better efficiency once the car is near or below top speed. It also helps keep the engine from over revving during highway driving or downhill segments.

5. Force and Acceleration (0-60 km/h Calculations)

1. **The driving force at the wheels is**

$$F = \frac{T_{\text{wheels}}}{r_{\text{tire}}},$$

2. **Force-Acceleration Table:**

Gear	Gear Ratio	Force	Acceleration
1st	3.237	19250N	11.1 m/s^2
2nd	2.104	12580N	7.23 m/s^2
3rd	1.422	8500N	4.88 m/s^2
4th	1.000	5970N	3.43 m/s^2
5th	0.814	4835N	2.78 m/s^2
6th	0.622	3700N	2.13 m/s^2

3. **Theoretical 0–100 km/h Time (Ignoring Shifts):**
- **Step -1 Reaching the first shift:** We start with the calculations for top speed at first gear losslessly(without loss factors of drag or aerodynamics). Which is 0 to 82.2 km/h or 0 to 22.83 m/s.
  - Time = t1 = Change in Velocity / Acceleration = 2.06 s for reaching 82.2 km/h in 1st gear.
- **Step - 2 Reaching 100 km/h after shifting to 2nd Gear:** Now the speed jump is $\Delta v \approx (27.78 - 22.83) = 4.95$ m/s. Acceleration a2nd ≈ 7.23 m/s^2 .
  - Time = t2 = 0.68s

$$T_{0\text{-}100} = t_1 + t_{shift} + t_2 \approx 2.06 + 0.2 + 0.68 \approx 2.94 \text{ s.}$$

Under lossless assumptions, ~ 2.9–3.0 s from 0–100 km/h is the theoretical floor.

6. Design Assumptions

1. **Steel Shafts**
   a. **AISI 4140** is a chromium-molybdenum alloy steel known for its high fatigue strength, toughness, and resistance to wear. After appropriate heat treatment, it exhibits:
      i. **Ultimate Tensile Strength (UTS):** Typically ranges from 655 to 1080 MPa, depending on the specific heat treatment process.
      ii. **Endurance Limit (σe):** Approximately 50% of the UTS, which translates to a range of 327.5 to 540 MPa. This makes it suitable for components subjected to cyclic loading, such as transmission shafts.
2. **Torsional Shock Factor**
   a. A **torsional shock factor** is introduced to ensure the shaft can handle these transient loads without yielding. A typical value ranges from 1.2 to 1.3, providing a safety margin against such events.
3. **Safety Factor for Shafts**
   a. For rotating shafts under combined bending and torsion, a safety factor between 1.5 and 2.0 is standard.
4. **Bearings**
   a. For automotive applications, an L10 life of 1000 to 2000 hours under peak load conditions is typical, aligning with the vehicle's expected service life.
   b. Equivalent bearing load is assumed as $P=X*F_r+Y*F_a$.
   c. Tapered roller bearings are assumed standard for combined radial + thrust loads.
5. **Lubrication**
   a. **Synthetic 75W-90 gear oil** is often used due to its stability across a wide temperature range and compatibility with synchronizer materials. The inclusion of **friction modifiers** aids in smooth gear engagement. Operating temperatures typically range from 80°C to 120°C, depending on driving conditions.
6. **Housing Material**
   a. **Cast aluminum** is favored for transmission housings because of its light weight and good thermal conductivity.
   b. Split-housing design for assembly convenience.

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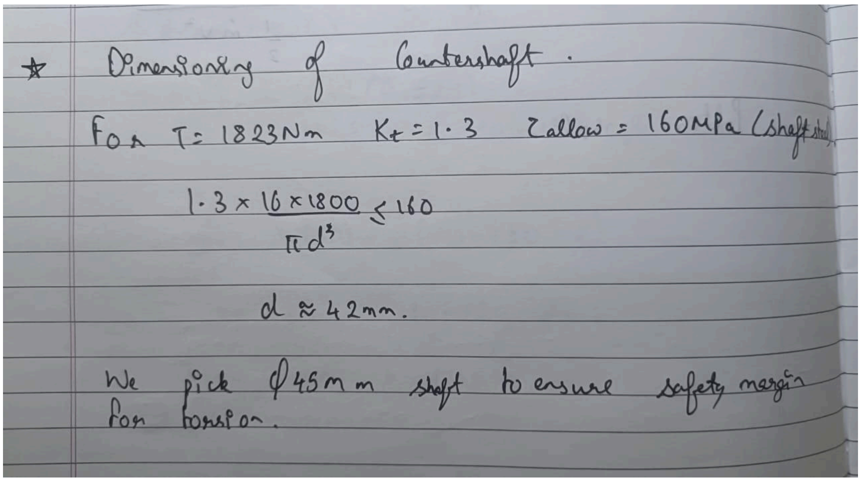
7. Shaft Analysis

1. Assumptions:
   a. K_0 = 1.25 - 1.5
   b. K_v = 1.1 - 1.2
   c. K_s = 1 - 1.1

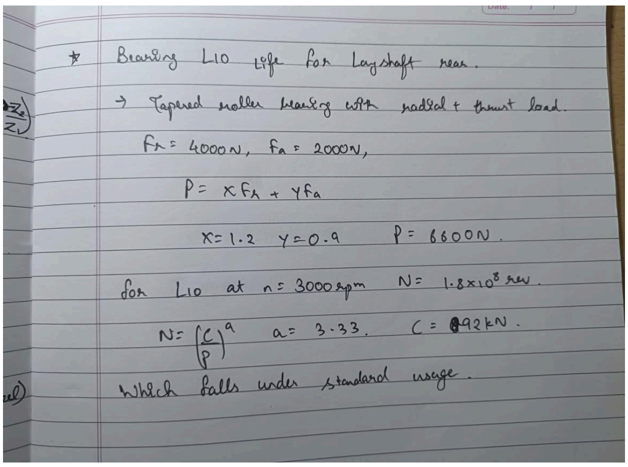
d. K_m = 1.1 - 1.2
e. K_t = 1.

2. **Torque and Stress Calculation:**
    a. From earlier torque multiplication:
        i. Max torque on input shaft ≈ 563 Nm,
        ii. Countershaft ≈ 563×3.237 ≈ 1823Nm (1st gear step).
    b. Peak torsional stress for a circular shaft (no keyway): $\tau_{\text{max,actual}} = K_t \, \frac{16\,T}{\pi\,d^3}.$
    c. We impose τmax,actual ≤ ασe, where α is a fraction of the endurance limit for torsional loading (commonly α ≈ 0.58 for shear). Suppose σe = 280MPa for our shaft steel in torsion, so ασe ≈ 160MPa
    d. The diameter of the countershaft is roughly 42 mm. We select 45 mm to account for safety margin.



8. Bearing Selection (ASME)

1. **Load Summation:** Fr,Fa from each gear pair.
2. **Equivalent Load:** P=X*F_r+Y*F_a
3. **L10 Life formula:** $L_{10} = \left(\frac{C}{P}\right)^a,$ **,** where a=3.33 for tapered rollers.
4. Bearing Material Selection & Reasoning–
    a. Typically a high-carbon bearing steel (SAE 52100 or equivalent).
        i. Hardness > 60HRC, good rolling contact fatigue life.
        ii. Standard for automotive tapered roller or ball bearings.
        iii. Good for L10 ~ 200k+ km design life
        iv. Calculations:



9. Key/Spline Selection:

While older designs used parallel keys, modern gearboxes use involute splines:

$$\tau_{\text{avg}} = \frac{2\,T}{d\,l_{\text{eng}} \times \text{No. of teeth}}.$$

If τavg < τallow, we accept the design.

1. Spline Material Selection & Reasoning– Material: Typically the same steel as the shaft (4140) or a gear-grade steel (8620) for case-hardening.
    a. Uniform heat treatment with the shaft for consistent core hardness.
    b. If the external gear is 8620, it might be carburized for high surface hardness, ensuring minimal wear at the spline engagement.
2. Calculations:

Spline Tooth Shear.

T = 1800 Nm with 6 teeth, engaged length ~25 mm
tooth thickness = 3 mm.

$F = \frac{T}{r}$: $r \approx 25$ m.

Shear area = 450 mm^2.
F = 72000 N.
τ_{avg} = 160 MPa.

FS ≈ 1.5. Acceptable ✓

10. Lubrication System (ASME)

1. Chosen Oil: Synthetic 75W-90 gear oil, GL-4 or GL-5 rating.
    a. Provides EP (extreme pressure) protection for gear flanks.
    b. Maintains stable viscosity up to ~ 120◦C.
    c. Common in performance transmissions with synchros.– Volume: ~2L for typical 6-speed manual.
2. Calculations for testing:

Oil Volume for Splash depth.

Ensuring partial submersion of the largest gear.

if C/s A = 0.04 m^2 & we want depth = 0.015 m

Volume = 0.6 L.

For heat capacity management we keep a total of 2L.

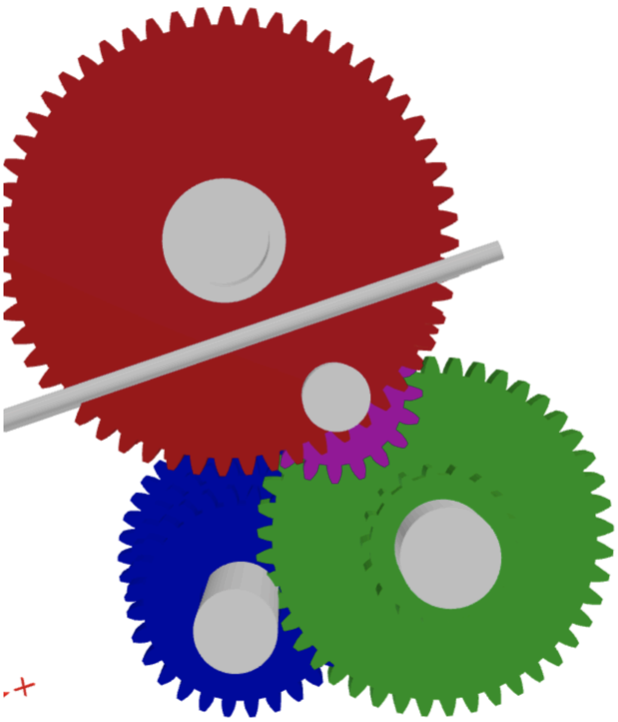
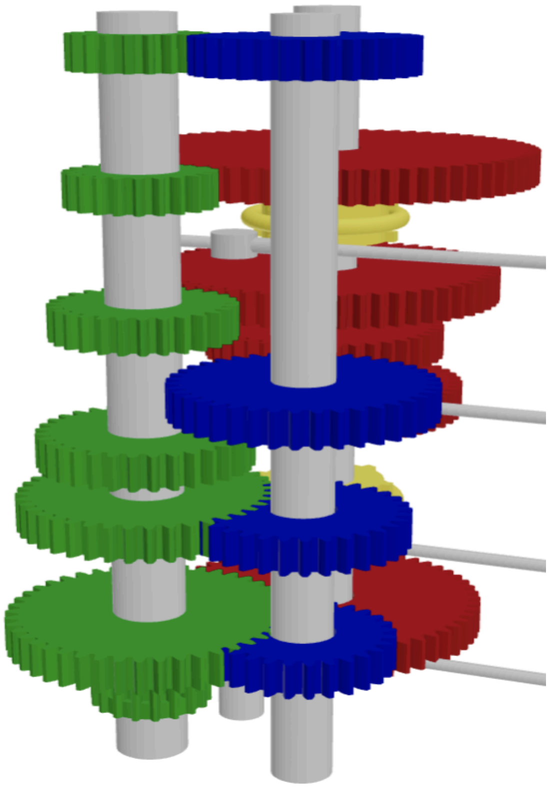
∴ layshaft gear stays partially submerged.

11. Housing Design and Assembly

1. **Housing Material**: Cast aluminum (e.g., Al-Si alloy or A356).
    a. Light weight, good thermal conductivity, standard for automotive transmissions.
    b. Enough ultimate strength (> 250MPa) and yields ~ 150MPa plus advantage of ribbing.
2. Wall Thickness: 3–5mm plus local bossing at bearing seats for minimal deflection.
3. Seals and Gaskets: Use radial lip seals around input/output shafts, a perimeter gasket or RTV for the main split.
4. **Assembly Layout**
    a. Input Shaft: Splined to the clutch, carrying the 4th gear (if 4th is direct drive).
    b. Countershaft (layshaft): Holds the cluster gears that remain in constant mesh with the main shaft gears
    c. Output (main) Shaft: Gears can freewheel until engaged by synchronizers/dog clutches.
    d. Shift Mechanism: Shift forks move the synchro hubs to lock a gear to the main shaft
    e. Oil Level: Typically near the bottom of the layshaft gear, ensuring partial immersion for splash lubrication

7. Assembly Discussion

We start the discussion by showing the assembly made on OpenScad :

Bottom View	Side View
	

1. Basic Details for the gears:
    a. Module = 2.

    b. Pressure Angle = 20
    c. Clearance = 0.5 mm
    d. Gear Width = 16 mm
2. Color coding of the gears and shafts:
    a. **All shafts** = Silver
    b. **Input Gear Color:** darkblue
    c. **Layshaft Gear Color:** dark green
    d. **Output Gear Color:** darkred
    e. **Reverse Gear Color:** purple (for both layshaft and idler)
1. **Shafts details**:

Shaft	Length (mm)	Diameter (mm)
Input Shaft	350	25
Layshaft	350	30
Output Shaft	350	28
Idler Shaft	350*	20

2. **Input Shaft gears:**

Gear Name	Teeth	Module	Pressure Angle (°)	Clearance (mm)	Gear Width (mm)
Input Drive	32	2.5	20	0.5	16
4th Gear	40	2.5	20	0.5	16
5th Gear	32	2.5	20	0.5	16
6th Gear	28	2.5	20	0.5	16

3. **LayShaft Gears:**

Gear Name	Teeth	Module	Pressure Angle (°)	Clearance (mm)	Gear Width (mm)
Drive Gear	20	2.5	20	0.5	16
1st Gear	22	2.5	20	0.5	16

2nd Gear	28	2.5	20	0.5	16
3rd Gear	33	2.5	20	0.5	16
5th Gear	40	2.5	20	0.5	16
6th Gear	45	2.5	20	0.5	16
Reverse Gear	18	2.5	20	0.5	16

4. **Output Shafts Gears:**

Gear Name	Teeth	Module	Pressure Angle (°)	Clearance (mm)	Gear Width (mm)
1st Gear	71	2.5	20	0.5	16
2nd Gear	59	2.5	20	0.5	16
3rd Gear	47	2.5	20	0.5	16
4th Gear	40	2.5	20	0.5	16
Reverse Gear	56	2.5	20	0.5	16

5. **Idler Shaft Gears:**

Gear Name	Teeth	Module	Pressure Angle (°)	Clearance (mm)	Gear Width (mm)
Reverse (Idler)	20	2.5	20	0.5	16

11. Conclusion:

This gearbox design report captures the essential aspects of sizing and selecting shafts, bearings, and gears for a high-performance 6-speed manual transmission. It recognizes both the theoretical and practical limits of speed, highlights the importance of robust material choices (AISI 4140 steel for shafts, SAE 52100 bearings), and provides a glimpse into the interplay between powertrain capabilities and aerodynamic constraints. The content thoroughly addresses many core design principles and presentation of gear tooth stresses, shaft deflection, thermal considerations with synchronizer design. Ultimately, **the design approach and references used** (AGMA, ISO, ASME, manufacturer catalogs) ensure that the gearbox is aligned with industry best practices.

12. References:

1. AGMA2001-D04, Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth.
2. AGMA 925-A03, Effect of Lubrication on Gear Surface Distress.
3. ISO 6336, Calculation of Load Capacity of Spur and Helical Gears.
4. ASME B106.1M, Design of Transmission Shafting, American Society of Mechanical Engineers.
5. Design and Analysis of Six Speed Gearbox by Ujjayan Majumdar, Sujit Maity, Gora Chand Chell
6. ASME B17.1, Keys and Keyseats.
7. J. E. Shigley, C. R. Mischke, and R. G. Budynas, Mechanical Engineering Design, McGraw-Hill.
8. N. K. Mehta, Machine Tool Design and Numerical Control, Tata McGraw-Hill.
9. T. D. Gillespie, Fundamentals of Vehicle Dynamics, SAE International.
10. SKF or Timken catalogs, Bearing Selection Guides with dynamic capacity references.